

MCGINN & GIBB, PLLC
A PROFESSIONAL LIMITED LIABILITY COMPANY
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8321 OLD COURTHOUSE ROAD, SUITE 200
VIENNA, VIRGINIA 22182-3817
TELEPHONE (703) 761-4100
FACSIMILE (703) 761-2375; (703) 761-2376

**APPLICATION
FOR
UNITED STATES
LETTERS PATENT**

APPLICANT'S: KOJI MATSUNO, ET AL.

**FOR: DIFFERENTIAL LIMITING CONTROL
APPARATUS FOR VEHICLE**

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1 TITLE OF THE INVENTION

2 DIFFERENTIAL LIMITING CONTROL APPARATUS FOR VEHICLE

3
4 BACKGROUND OF THE INVENTION

5 1. Field of the invention

6 The present invention relates to a differential
7 limiting control apparatus provided in a center differential of
8 a four wheel drive vehicle for performing a differential limiting
9 control between a front drive shaft on the front wheel side and
10 a rear drive shaft on the rear wheel side.

11 2. Discussion of related arts

12 In general, many of power transmission controls such
13 as front-rear wheels power distribution controls or front-rear
14 /left-right wheels differential limiting controls, variably
15 control the clutch engagement force by a multiple disc clutch
16 and the like. With respect to the differential limiting control
17 in which the clutch engagement force is variably controlled, the
18 vehicle driving performance is largely dependant on the control
19 characteristics of the differential limiting control. Further,
20 the vehicle driving performance varies according to tires or road
21 surface conditions with the same control logics and control
22 constants. Further, the required vehicle driving performance
23 differs with an individual driver or with driving conditions.

24 Hence, Japanese Patent Application Laid-open
25 Toku-Kai-Hei 8-132914 discloses a technology of a front-rear

1 wheels torque distribution control apparatus in which a driver
2 establishes a differential limiting torque by the manual operation
3 to obtain a desired torque distribution condition.

4 In order to realize the driving performance of the
5 vehicle as a driver desires, however, it is necessary to variably
6 control the differential limiting torque according to a variety
7 of driving conditions or road surface conditions. Therefore, it
8 is difficult to adjust the differential limiting torque to a target
9 value and to realize the most suitable driving performance of
10 the vehicle merely by establishing the differential limiting
11 torque directly by the manual operation of the driver, as proposed
12 in Toku-Kai-Hei 8-132914. Further, in case where the driver
13 manually operates the differential limiting torque control
14 apparatus and drives the vehicle, for example, in a released
15 condition of the differential limiting clutch as the driver
16 intends, in an extreme case, a spin may occur to the vehicle on
17 a road surface with low friction coefficient. Reversely, in case
18 where the driver manually operates the differential limiting
19 torque to drive the vehicle in an engaged condition of the
20 differential limiting clutch for a long time, sometimes wrong
21 loads such as an internal circulation torque occurs to the
22 powertrain of the vehicle. Further, such wrong loads may
23 exacerbate fuel economy.

24

25 SUMMARY OF THE INVENTION

1 It is an object of the invention to provide a differential
2 limiting control apparatus capable of realizing a natural driving
3 performance of a vehicle suitable for driving conditions.

4 In order to attain the object, a differential
5 limiting control apparatus for a four wheel drive vehicle having
6 clutch means for variably transmitting a driving force to a front
7 drive shaft and to a rear drive shaft, comprises automatic clutch
8 control means for automatically calculating and establishing an
9 engagement force of the clutch means according to traveling
10 conditions of the vehicle, manual clutch control means for manually
11 establishing the engagement force of the clutch means and control
12 selecting means for selecting either of the automatic clutch
13 control means and the manual clutch control means and for commanding
14 the selected one to output the engagement force.

15 In an initial condition of an ignition switch turned
16 on, the control selecting means command the automatic clutch
17 control means to output the engagement force of the clutch means
18 until the manual clutch control means is newly selected.

19 Further, when the vehicle travels at a higher speed
20 than a threshold value, the control selecting means command the
21 automatic clutch control means to output the engagement force
22 of the clutch means.

23

24 BRIEF DESCRIPTION OF THE DRAWINGS

25 Fig. 1 is a schematic drawing showing a vehicle power

1 train and a center differential incorporating a differential
2 limiting control section;

3 Fig. 2 is a functional block diagram of the differential
4 limiting control section;

5 Fig. 3 is a functional block diagram of an automatic
6 mode control section;

7 Fig. 4 is an explanatory diagram of an example of a
8 table showing a relationship between a vehicle speed and a control
9 start differential rotation speed;

10 Fig. 5 is an explanatory diagram of an example of a table
11 showing a relationship between a lateral acceleration and a
12 correction coefficient of the control start differential rotation
13 speed; and

14 Fig. 6 is a flowchart of a front-rear wheels power
15 distribution control program.

16

17 DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

18 Referring now to Fig. 1, reference numeral 1 denotes
19 an engine mounted on a front part of a vehicle. Driving force
20 of the engine 1 is transmitted to a center differential 3 through
21 an automatic transmission 2 (including a torque converter) and
22 a transmission output shaft 2a. Further, the driving force of
23 the engine 1 inputs from the center differential 3 to a rear final
24 reduction gear unit 7 through a rear drive shaft 4, a propeller
25 shaft 5 and a drive pinion 6 and on the other hand the driving

1 force inputs from the center differential 3 to a front final
2 reduction gear unit 11 through a transfer drive gear 8, a transfer
3 driven gear 9 and a front drive shaft 10. The automatic
4 transmission 2 is accommodated integrally with the center
5 differential 3 and the front final reduction gear unit 11 in a
6 casing 12.

7 The driving force inputted to the rear final reduction
8 gear unit 7 is transmitted to a rear left wheel 14RL and a rear
9 right wheel 14RR through a rear left drive shaft 13RL and a rear
10 right drive shaft 13RR, respectively. Further, the driving force
11 inputted to the front final reduction gear unit 11 is transmitted
12 to a front left wheel 14FL and a front right wheel 14FR through
13 a front left axle shaft 13FL and a front right axle shaft 13FR,
14 respectively.

15 The center differential 3 incorporates a first sun gear
16 15 having a large diameter and mounted on the transmission output
17 shaft 2a. The first sun gear 15 meshes with a first pinion 16
18 having a small diameter, thus a first gear train being constituted.

19 Further, a second sun gear 17 having a small diameter
20 is mounted on the rear drive shaft 4 from which power is transmitted
21 to rear wheels and meshes with a second pinion 18 having a large
22 diameter, thus a second gear train being constituted.

23 The first pinion 16 and the second pinion 18 are
24 integrally formed with a pinion member 19 which is rotatably
25 supported by a fixed shaft provided in a carrier 20. Further,

1 the carrier 20 is connected at the front thereof with the transfer
2 drive gear 8 from which power is transmitted to the front wheels.

3 Further, the carrier 20 is rotatably fitted at the front
4 section thereof over the output shaft 2a of the transmission 2
5 and is rotatably fitted at the rear section thereof over the rear
6 drive shaft 4. Further, the first and second sun gears 15, 17
7 are accommodated in the central space of the carrier 20. In Fig.
8 1, only one pinion member 19 is illustrated, however in an actual
9 construction, plural pinion members 19 are provided around the
10 sun gears 15, 17.

11 Thus, the center differential 3 is formed as a compound
12 planetary gear unit having an input member in the transmission
13 shaft 2a, an output member in the rear drive shaft 4 and the other
14 output member in the carrier 20.

15 The center differential 3 of a compound planetary type
16 is provided with a differential function by properly establishing
17 the number of teeth of the first and second sun gears 15, 17 and
18 the first and second pinions 16, 18.

19 Further, the center differential 3 is furnished with
20 a desired base torque distribution, for example an unequal torque
21 distribution biased on rear wheels, by appropriately establishing
22 working pitch circles of the first and second sun gears 15, 17
23 and the first and second pinions 16, 18.

24 Further, the center differential 3 is designed in such
25 a manner that the first and second sun gears 15, 17 and the first

1 and second pinions 16, 18 have helical teeth, respectively,
2 leaving thrust loads. As a result, the thrust loads produce a
3 friction torque at an end of the respective pinion members 19.
4 Further, a resultant force of separation force and tangential
5 force generated by meshing of the gears exerts on the fixed shaft
6 provided in the carrier 20, producing another friction torque
7 between the respective pinion members 19 and the respective fixed
8 shafts. Since these friction torques are obtained as a
9 differential limiting torque which is proportional to the input
10 torque, a differential limiting function is given to the center
11 differential 3 itself.

12 Further, there is provided a center differential clutch
13 (transfer clutch) 21 of a hydraulic multiple disc clutch type
14 for varying the front-rear torque distribution between two
15 output members, the carrier 20 and the rear drive shaft 4, of
16 the center differential 3. By controlling the engagement force
17 of this transfer clutch 21, the front-rear torque distribution
18 ratio can be varied from 50:50 in a fully engaged condition to
19 an inherent front-rear torque distribution ratio, for example
20 35:65, of the center differential 3 in a released condition.

21 The transfer clutch 21 is connected with a center
22 differential clutch drive section 60 constituted by a hydraulic
23 circuit including a plurality of solenoid valves. Hydraulic
24 pressure generated in the center differential clutch drive section
25 60 actuates a piston (not shown) to engage or release the transfer

1 clutch 21. Further, control signals for driving the center
2 differential clutch drive section 60, that is, input signals to
3 the respective solenoid valves, are outputted from a differential
4 limiting control section 50 which will be described hereinafter.

5 The rear final reduction gear unit 7 comprises a
6 differential mechanism 22 using bevel gears and a rear differential
7 clutch 23 using a multiple disc clutch. The rear differential
8 clutch 23 is provided between a differential case 25 to which
9 a ring gear 24 is fixed and a rear right axle shaft 13RR. The
10 ring gear 24 meshes with the drive pinion 6 to drive the differential
11 mechanism 22.

12 The front final reduction gear unit 11 is constituted
13 by a differential mechanism 26 of bevel gear type and a front
14 disc clutch 27 using multiple discs in the same manner as the
15 rear final reduction gear unit 7. The front disc clutch 27 is
16 provided between a differential case 29 to which a ring gear 28
17 is fixed and a front right axle shaft 13FR. The ring gear 28 meshes
18 with a drive pinion of the front drive shaft 10 to drive the
19 differential mechanism 26.

20 The differential limiting control section 50 inputs
21 parameters necessary for controls from respective sensors and
22 switches. Wheel speeds of the wheels, 14FL, 14FR, 14RL and 14RR
23 are detected by wheel speed sensors 31FL, 31FR, 31RL and 31RR,
24 respectively and are inputted to the differential limiting
25 control section 50. Further, the differential limiting control

1 section 50 inputs a lateral acceleration Gy applied to the vehicle
2 from a lateral acceleration sensor 32. Also, the differential
3 limiting control section 50 inputs ON-OFF signals from a brake
4 switch 33 provided in the vehicle. The brake switch 33 outputs
5 a turned-on signal when a brake pedal (not shown) is depressed
6 and a turned-off signal when the brake pedal is eased up. Further,
7 the differential limiting control section 50 inputs ON-OFF signals
8 from an ignition switch 34. The vehicle incorporates a mode switch
9 35 for selecting an automatic mode in which the front-rear power
10 distribution control is performed automatically according to the
11 driving conditions of the vehicle or a manual mode in which the
12 front-rear power distribution control is performed manually
13 according to a driver's intention and the differential limiting
14 control section 50 inputs the selection signal from the mode switch
15 35. When the driver selects the manual mode, the engagement
16 condition of the transfer clutch 21 is freely selected from a
17 released condition to a fully engaged condition by the driver's
18 operation of a characteristic changing dial 36. Further, the mode
19 presently selected is indicated by a mode indicator lamp 37 provided
20 on the instrument panel. Further, the vehicle incorporates a known
21 antilock braking system (ABS) for preventing wheel locks on
22 applying brakes and an operation signal of the ABS is outputted
23 from an ABS control apparatus 38 to the differential limiting
24 control section 50.

25 The differential limiting control section 50 is

1 constituted by a micro-computer and peripheral circuits,
2 specifically, as shown in Fig. 2, constituted by a vehicle speed
3 calculating section 51, an automatic mode control section 52,
4 a manual mode control section 53, a mode establishing section
5 54 and a clutch torque calculating section 55.

6 The vehicle speed calculating section 51 inputs wheel
7 speeds ω_{fl} , ω_{fr} , ω_{rl} , ω_{rr} of the wheels 14FL, 14FR, 14RL, 14RR
8 from the wheel speed sensors 31FL, 31FR, 31RL, 31RR, respectively.
9 A vehicle speed V is calculated by averaging these wheel speeds
10 and is outputted to the automatic mode control section 52 and
11 the mode establishing section 54.

12 The automatic mode control section 52 inputs wheel
13 speeds ω_{fl} , ω_{fr} , ω_{rl} , ω_{rr} of the wheels 14FL, 14FR, 14RL, 14RR
14 from the wheel speed sensors 31FL, 31FR, 31RL, 31RR, respectively.
15 Further, the automatic mode control section 52 inputs the lateral
16 acceleration G_y from the lateral acceleration sensor 32, the
17 braking signal from the braking switch 33, the vehicle speed V
18 from the vehicle speed calculating section 51 and an output signal
19 indicative of the calculation value from the mode establishing
20 section 54.

21 The automatic mode control section 52 acts as automatic
22 clutch control means and, more specifically, calculates target
23 front-rear differential rotation speeds (target differential
24 rotation speeds between front and rear drive shafts) $\Delta\omega_{ctrft}$,
25 $\Delta\omega_{ctrtrt}$ which will be described hereinafter, a target front

1 left-right differential rotation speed (target differential
 2 rotation speed between front left and front right wheels) $\Delta \omega$
 3 F_{tt} , and a target rear left-right differential rotation speed
 4 (target differential rotation speed between rear left and rear
 5 right wheels) $\Delta \omega R_{rt}$ and calculates actual front-rear
 6 differential rotation speeds (actual differential rotation speeds
 7 between front and rear wheels) $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrr}$, an actual front
 8 left-right differential rotation speed (actual differential
 9 rotation speed between front left and front right wheels) $\Delta \omega$
 10 F_t , and an actual rear left-right differential rotation speed
 11 (actual differential rotation speed between rear left and rear
 12 right wheels) $\Delta \omega R_r$. Then, those respective deviations ε_{ctrf} ,
 13 ε_{ctrr} , ε_{Ft} , ε_{Rr} are calculated respectively. Further, a
 14 switching function being constituted using the polarity of the
 15 integral term of these deviations, a clutch torque T_{lsdfb} is
 16 calculated by applying the sliding mode control and further by
 17 taking a deviation proportional part into consideration. Thus
 18 calculated clutch torque T_{lsdfb} is outputted to the clutch torque
 19 calculating section 55 when the mode establishing section 54 inputs
 20 a command for outputting the calculation value.

21 That is, as shown in Fig. 3, the automatic mode control
 22 section 52 is constituted by a brake switch delaying section 52a,
 23 an actual front-rear differential rotation speed calculating
 24 section 52b, an actual front left-right differential rotation
 25 speed calculating section 52c, an actual rear left -right

1 differential rotation speed calculating section 52d, a control
2 start differential rotation speed calculating section 52e, a
3 target differential rotation speed establishing section 52f, a
4 sliding mode control clutch torque calculating section 52g, a
5 deviation proportional control clutch torque calculating
6 section 52h, and a feedback control clutch torque calculating
7 section 52i.

8 The brake switch delaying section 52a acts as delaying
9 the timing for changing over the brake switch 33 from a turned-
10 on condition to a turned-off condition for a specified short time.
11 That is, when the brake switch 33 is changed over from a turned-
12 on condition to a turned-off condition, the brake switch 33 is
13 not turned off until the specified time elapses. Such a delaying
14 process is not performed, when the brake switch 33 is changed
15 over from a turned-off condition to a turned-on condition. After
16 the specified time elapses, the brake switch 33 outputs an OFF
17 signal to the sliding mode control clutch torque calculating
18 section 52g and the deviation proportional control clutch torque
19 calculating section 52h.

20 The actual front-rear differential rotation speed
21 calculating section 52b inputs wheel speeds ω_{fl} , ω_{fr} , ω_{rl} , ω_{rr}
22 of the wheels 14FL, 14FR, 14RL, 14RR from the wheel speed
23 sensors 31FL, 31FR, 31RL, 31RR, respectively. Based on these wheel
24 speeds, two kinds of the actual front-rear differential rotation
25 speeds, $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrr}$ are calculated according to the following

1 formulas (1) and (2):

$$2 \quad \Delta \omega_{ctrf} = ((\omega_{fl} + \omega_{fr}) - (\omega_{rl} + \omega_{rr})) / 2 \quad (1)$$

$$3 \quad \Delta \omega_{ctrr} = ((\omega_{rl} + \omega_{rr}) - (\omega_{fl} + \omega_{fr})) / 2 \quad (2)$$

4 In case where the rotation speed of the front drive
5 shaft is faster than that of the rear drive shaft, $\Delta \omega_{ctrf}$ is
6 positive and $\Delta \omega_{ctrr}$ is negative. On the other hand, in case where
7 the rotation speed of the front drive shaft is slower than that
8 of the rear drive shaft, $\Delta \omega_{ctrf}$ is negative and $\Delta \omega_{ctrr}$ is
9 positive. Thus calculated actual front-rear differential rotation
10 speeds $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrr}$ are outputted to the sliding mode control
11 clutch torque calculating section 52g and the deviation
12 proportional control clutch torque calculating section 52h. The
13 reason why such two kinds of the actual front-rear differential
14 rotation speeds are calculated is that the clutch torque is
15 established such that the torque is transmitted from the drive
16 shaft whose rotation speed is fast to the shaft whose rotation
17 speed is slow by judging the positive or negative sign of the
18 actual front-rear differential rotation speeds $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrr}$.

19 The actual front left-right differential rotation speed
20 calculating section 52c inputs the wheel speeds ω_{fl} , ω_{fr} of the
21 front left and front right wheels 14FL, 14FR from the wheel speed
22 sensors 31FL, 31FR of the front left and front right wheels 14FL,
23 14FR and further inputs the lateral acceleration G_y from the lateral
24 acceleration sensor 32. An actual differential rotation speed
25 $\Delta \omega_{Ft}$ between the front left wheel 14FL and the front right wheel

1 14FR is calculated according to either of the following formulas
2 (3), (4) and (5):

3 $\Delta \omega_{Ft} = \omega_{fr} - \omega_{fl}$ (when turning right) (3)

4 $\Delta \omega_{Ft} = \omega_{fl} - \omega_{fr}$ (when turning left) (4)

5 $\Delta \omega_{Ft} = |\omega_{fr} - \omega_{fl}|$ (when traveling straight) (5)

6 Then, the turning condition of the vehicle is judged from the
7 lateral acceleration G_y . Further, when the absolute value $|G_y|$
8 is smaller than a preestablished value A_{ys} , it is judged that
9 the vehicle travels almost straightforward. In case where the
10 lateral acceleration G_y is larger than the preestablished value
11 A_{ys} , it is judged that the vehicle turns left and in case where
12 the lateral acceleration G_y is smaller than $-A_{ys}$, it is judged
13 that the vehicle turns right. The turning condition of the vehicle
14 can be judged from a yaw rate, a steering angle and the like.
15 When both left and right wheels have no wheel spin, the outer
16 wheel rotates faster than the inner wheel, therefore, the actual
17 differential rotation speed $\Delta \omega_{Ft}$ obtained from the formulas (3),
18 (4) has a negative sign. Thus calculated actual differential
19 rotation speed $\Delta \omega_{Ft}$ between the front left wheel 14FL and the
20 front right wheel 14FR is outputted to the sliding mode control
21 clutch torque calculating section 52g and the deviation
22 proportional control clutch torque calculating section 52h,
23 respectively.

24 The actual rear left-right differential rotation speed
25 calculating section 52d inputs the wheel speeds ω_{rl} , ω_{rr} of the

1 rear left and rear right wheels 14RL, 14RR from the wheel speed
2 sensors 31RL, 31RR of the rear left and rear right wheels 14RL,
3 14RR and inputs the lateral acceleration G_y from the lateral
4 acceleration sensor 32. An actual differential rotation speed
5 $\Delta \omega_{Rt}$ between the rear left wheel 14RL and the rear right wheel
6 14RR is calculated according to either of the following formulas
7 (6), (7) and (8):

$$8 \quad \Delta \omega_{Rt} = \omega_{rr} - \omega_{rl} \text{ (when turning right)} \quad (6)$$

$$9 \quad \Delta \omega_{Rt} = \omega_{rl} - \omega_{rr} \text{ (when turning left)} \quad (7)$$

$$10 \quad \Delta \omega_{Rt} = |\omega_{rr} - \omega_{rl}| \text{ (when traveling straight)} \quad (8)$$

11 Then, the turning condition of the vehicle is judged from the
12 lateral acceleration G_y . Further, when the absolute value $|G_y|$
13 is smaller than a preestablished value A_{ys} , it is judged that
14 the vehicle travels almost straightforward. Similarly to the
15 actual front left-right differential rotation speed calculating
16 section 50e, when the lateral acceleration G_y is larger than the
17 preestablished value A_{ys} , it is judged that the vehicle turns
18 left and when the lateral acceleration G_y is smaller than $-A_{ys}$,
19 it is judged that the vehicle turns right. The turning condition
20 of the vehicle may be judged from a yaw rate, a steering angle
21 and the like. When both left and right wheels have no wheel spin,
22 the outer wheel rotates faster than the inner wheel, therefore,
23 the actual differential rotation speed $\Delta \omega_{Rt}$ obtained from the
24 formulas (6), (7) has a negative sign. Thus calculated actual
25 differential rotation speed $\Delta \omega_{Rt}$ between the rear left wheel

1 14RL and the rear right wheel 14RR is outputted to the sliding
2 mode control clutch torque calculating section 52g and the
3 deviation proportional control clutch torque calculating section
4 52h, respectively.

5 The actual front-rear differential rotation speed
6 calculating section 52b, the actual front left-right differential
7 rotation speed calculating section 52c and the actual rear
8 left-right differential rotation speed calculating section 52d
9 function as actual differential rotation speed detecting means.

10 The control start differential rotation speed
11 calculating section 52e inputs the lateral acceleration G_y from
12 the lateral acceleration sensor 32 and the vehicle speed V from
13 the vehicle speed calculating section 51, respectively. Further,
14 a lower limit value of the actual differential rotation speed
15 between front and rear drive shafts $\Delta \omega_{ctrfb}$, that is, a front-rear
16 control start differential rotation speed (control start
17 differential rotation speed between front and rear drive shafts)
18 $\Delta \omega_{ctrfs}$ is established according to the vehicle speed V and the
19 lateral acceleration G_y by reference to a preestablished table.
20 Similarly, a lower limit value of the actual differential rotation
21 speed $\Delta \omega_{ctr r}$ between front and rear shafts, that is, a front-rear
22 control start differential rotation speed (control start
23 differential rotation speed between front and rear drive shafts)
24 $\Delta \omega_{ctr rs}$ is established according to the vehicle speed V and the
25 lateral acceleration G_y by reference to a preestablished table.

1 Further, a lower limit of the actual differential rotation speed
2 $\Delta \omega_{Ft}$ between the front left wheel 14FL and the front right wheel
3 14FR, that is, a control start front left-right differential
4 rotation speed $\Delta \omega_{Fts}$ is established to a constant value C_{Fts} .
5 Similarly, a lower limit of the actual differential rotation speed
6 $\Delta \omega_{Rt}$ between the rear left wheel 14RL and the rear right wheel
7 14RR, that is, a control start rear left-right differential
8 rotation speed $\Delta \omega_{Rrs}$ is established to a constant value C_{Rrs} .

9 In establishing the aforesaid front-rear control
10 start differential rotation speed $\Delta \omega_{ctrfs}$, first, as shown in
11 Fig. 4, a basic value $\Delta \omega_{ctrfsb}$ of the front-rear control start
12 differential rotation speed $\Delta \omega_{ctrfs}$ is established based on the
13 present vehicle speed V by referring to a basic table of the
14 front-rear control start differential rotation speed $\Delta \omega_{ctrfs}$
15 versus the vehicle speed V . Further, as shown in Fig. 5, a correction
16 coefficient $k_{\omega gy}$ of the front-rear control start differential
17 rotation speed $\Delta \omega_{ctrfs}$ is obtained based on the present lateral
18 acceleration G_y from a correction coefficient table showing the
19 relationship of the front-rear control start differential rotation
20 speed $\Delta \omega_{ctrfs}$ versus the lateral acceleration G_y . Then, the basic
21 front-rear control start differential rotation speed $\Delta \omega_{ctrfsb}$
22 is multiplied by the correction coefficient $k_{\omega gy}$ to obtain a final
23 front-rear control start differential rotation speed $\Delta \omega_{ctrfs}$
24 $(= \Delta \omega_{ctrfsb} \cdot k_{\omega gy})$.

25 According to a table of Fig. 4, the front-rear control

1 start differential rotation speed $\Delta \omega_{ctrfs}$ is established to be
2 larger with an increase of the vehicle speed V . This is why the
3 engagement force of the clutch is alleviated at high speeds for
4 the purpose of improving fuel economy. Further, according to a
5 table of Fig. 5, the front-rear control start differential rotation
6 speed $\Delta \omega_{ctrfs}$ is established to be larger with an increase of
7 the lateral acceleration G_y . This is why the engagement force
8 of the clutch is alleviated at high speeds to enhance the turning
9 ability of the vehicle.

10 The front-rear control start differential rotation
11 speed $\Delta \omega_{ctrfs}$ is established in the same manner as the front-rear
12 control start differential rotation speed $\Delta \omega_{ctrfs}$.

13 As will be described hereinafter, these respective
14 control start differential rotation speeds $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctrfs}$,
15 $\Delta \omega_{Fts}$ and $\Delta \omega_{Rrs}$ are threshold values for starting the
16 differential limiting control between the front and rear shafts,
17 between the front left and front right wheels and between the
18 rear left and rear right wheels. In case where the actual
19 differential rotation speeds $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrf}$, $\Delta \omega_{Ft}$ and $\Delta \omega$
20 Rr are smaller than the control start differential rotation speeds
21 $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctrfs}$, $\Delta \omega_{Fts}$ and $\Delta \omega_{Rrs}$, the engagement torque of
22 the transfer clutch 21 is established to 0. In particular, in
23 case where the front-rear differential rotation speed actually
24 controlled becomes so small that miscellaneous troubles such as
25 sticking of the transfer clutch 21 in a static friction condition,

1 delaying the convergence of the control due to the transfer clutch
2 21 in a slip-lock condition or exacerbating the control stability,
3 are caused. Further, in case where the control start differential
4 rotation speeds $\Delta \omega_{Fts}$, $\Delta \omega_{Rrs}$ between the front left and front
5 right wheel or between the rear left and rear right wheel are
6 set to 0 for example, when the wheel speed of the inner wheel
7 is greater than that of the outer wheel, immediately the
8 differential limiting control of the center differential is
9 carried out.

10 In this embodiment, the control start front-rear
11 differential rotation speeds $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctr rs}$ is established
12 according to the vehicle speed V and the lateral acceleration
13 G_y , however, those may be established according to a torque inputted
14 to the center differential 3 (center differential input torque).
15 Further, the control start front-rear differential rotation speeds
16 $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctr rs}$ may be selectively variable according to a
17 driver's intention.

18 Further, according to this embodiment, the control
19 start front left-right differential rotation speed $\Delta \omega_{Fts}$ and
20 the control start rear left-right differential rotation speed
21 $\Delta \omega_{Rrs}$ are established to the constant value, however, those may
22 be variably established according to parameters showing vehicle
23 behaviors.

24 Thus calculated control start differential rotation
25 speeds $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctr rs}$, $\Delta \omega_{Fts}$, $\Delta \omega_{Rrs}$ are outputted to the

1 target differential rotation speed establishing section 52f, the
2 sliding mode control clutch torque calculating section 52g and
3 the deviation proportional control clutch torque calculating
4 section 52h, respectively.

5 The target differential rotation speed establishing
6 section 52f inputs the respective control start differential
7 rotation speeds $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctr rs}$, $\Delta \omega_{Fts}$, $\Delta \omega_{Rrs}$ from the
8 control start differential rotation speed calculating section
9 52e and calculates target front-rear differential rotation
10 speeds $\Delta \omega_{ctrft}$, $\Delta \omega_{ctr rt}$, a target front differential rotation
11 speed $\Delta \omega_{Ftt}$, a target rear differential rotation speed $\Delta \omega_{Rrt}$
12 according to the following formulas (9), (10), (11), (12):

$$13 \quad \Delta \omega_{ctrft} = \Delta \omega_{ctrfs} + C_{ctrft} \quad (9)$$

$$14 \quad \Delta \omega_{ctr rt} = \Delta \omega_{ctr rs} + C_{ctr rt} \quad (10)$$

$$15 \quad \Delta \omega_{Ftt} = \Delta \omega_{Fts} + C_{Ftt} \quad (11)$$

$$16 \quad \Delta \omega_{Rrt} = \Delta \omega_{Rrs} + C_{Rrt} \quad (12)$$

17 where C_{ctrft} , $C_{ctr rt}$, C_{Ftt} , C_{Rrt} are constants established
18 beforehand based on calculations, experiments and the like. Thus
19 calculated target differential rotation speeds, $\Delta \omega_{ctrft}$, $\Delta \omega_{ctr rt}$,
20 $\Delta \omega_{Ftt}$, $\Delta \omega_{Rrt}$ are outputted to the sliding mode control
21 clutch torque calculating section 52g and the deviation
22 proportional control clutch torque calculating section 52h. Thus,
23 the control start differential rotation speed calculating section
24 52e and the target differential rotation speed establishing
25 section 52f serve as target differential rotation speed

1 establishing means in the automatic clutch control means.

2 The sliding mode control clutch torque calculating
3 section 52g inputs the brake signal processed by the brake switch
4 delaying section 52a, the actual differential rotation speeds
5 between the front and rear drive shafts $\Delta \omega_{ctrf}$, $\Delta \omega_{ctrtrt}$ from
6 the actual front-rear differential rotation speed calculating
7 section 52b, the actual differential rotation speed $\Delta \omega_{Ft}$ between
8 the front left wheel 14FL and the front right wheel 14FR from
9 the actual front left-right differential rotation speed
10 calculating section 52c, the actual differential rotation speed
11 $\Delta \omega_{Rr}$ between the rear left wheel 14RL and the rear right wheel
12 14RR from the actual rear left-right differential rotation speed
13 calculating section 52d, the respective control start differential
14 rotation speeds $\Delta \omega_{ctrfs}$, $\Delta \omega_{ctrtrs}$, $\Delta \omega_{Fts}$, $\Delta \omega_{Rrs}$ from the
15 control start differential rotation speed establishing section
16 52e, and the respective target differential rotation speeds Δ
17 ω_{ctrft} , $\Delta \omega_{ctrtrt}$, $\Delta \omega_{Ftt}$, $\Delta \omega_{Rrt}$ from the target differential
18 rotation speed establishing section 52f. Further, the sliding
19 mode control clutch torque calculating section 52g calculates
20 the deviations of the respective rotation speeds between the target
21 differential rotation speeds and the actual differential rotation
22 speeds. Then, the sliding mode control clutch torque calculating
23 section 52g constituting a switching function using the polarity
24 of the integral term of these deviations, calculates the clutch
25 torque of the transfer clutch 21 by applying the sliding mode

1 control.

2 The deviations of the respective rotation speeds between
3 the target differential rotation speeds and the actual
4 differential rotation speeds are calculated as follows:

$$5 \quad \varepsilon_{ctrf} = \Delta \omega_{ctrf} - \Delta \omega_{ctrft} \quad (13)$$

$$6 \quad \varepsilon_{ctrr} = \Delta \omega_{ctrr} - \Delta \omega_{ctrrt} \quad (14)$$

$$7 \quad \varepsilon_{Ft} = \Delta \omega_{Ft} - \Delta \omega_{Ftt} \quad (15)$$

$$8 \quad \varepsilon_{Rr} = \Delta \omega_{Rr} - \Delta \omega_{Rrt} \quad (16)$$

9 where ε_{ctrf} is a differential rotation speed deviation between
10 front and rear drive shafts; ε_{ctrr} is a differential rotation
11 speed deviation between front and rear drive shafts; ε_{Ft} is a
12 differential rotation speed deviation between front left and front
13 right wheels; and ε_{Rr} is a differential rotation speed deviation
14 between rear left and rear right wheels.

15 Further, the clutch torques $TSMC_{ctrf}$, $TSMC_{ctrr}$, $TSMC_{Ft}$,
16 $TSMC_{Rr}$ to be exerted on the transfer clutch 21 by the sliding
17 mode control for each rotation speed are calculated using the
18 following changing functions (17), (20), (23), (26).

19 First, the establishment of the clutch torque $TSMC_{ctrf}$
20 using the differential rotation speed deviation between front
21 and rear drive shafts ε_{ctrf} by the sliding mode will be described.

$$22 \quad TSMC_{ctrf} = s a t (x_{ctrf}) \quad (17)$$

23 where, in case of $x_{ctrf} > 0$, $TSMC_{ctrf} = s a t (x_{ctrf}) = x_{ctrf}$;

24 in case of $x_{ctrf} \leq 0$, $TSMC_{ctrf} = s a t (x_{ctrf}) = 0$;

$$25 \quad x_{ctrf} = k_{wctrf} \cdot J_w \cdot (d\varepsilon_{ctrf}/dt)$$

$$1 \quad + T_{sg} \cdot (s_{ctrf} / (|s_{ctrf}| + \delta)) \quad (18)$$

$$2 \quad \text{where } s_{ctrf} = \varepsilon_{ctrf} + k_i \cdot \int (\varepsilon_{ctrf}) dt \quad (19)$$

3 (limits of integral is from 0 to t);

4 k_{wctrf} is a differential term gain and in case of $(d\varepsilon_{ctrf}/dt) > 0$
 5 is k_{wu} and in case of $(d\varepsilon_{ctrf}/dt) \leq 0$, is k_{wd} ; J_w is an inertia
 6 term; T_{sg} is a changing gain; δ is a constant for preventing
 7 chatterings; and k_i is an integral term gain.

8 In case where the actual differential rotation speed
 9 $\Delta \omega_{ctrf}$ becomes smaller than the control start differential
 10 rotation speed $\Delta \omega_{ctrfs}$, the transfer clutch 21 is engaged in
 11 a static friction condition due to the too small differential
 12 rotation speed and as a result the control of the transfer clutch
 13 21 enters into a slip-lock condition. To prevent this exacerbated
 14 control stability, the clutch torque $T_{SMCctrf}$ is established to
 15 0. Further, the integral is reset ($\int (\varepsilon_{ctrf}) dt$ is established
 16 to 0). Further, in case where the brake ON signal is inputted,
 17 similarly, the clutch torque $T_{SMCctrf}$ is established to 0 in order
 18 to prevent the interference with the braking condition and also
 19 the integral is reset.

20 Next, the establishment of the clutch torque T_{SMCctr}
 21 using the differential rotation speed deviation between front
 22 and rear drive shafts ε_{ctr} by the sliding mode will be described.

$$23 \quad T_{SMCctr} = s a t (x_{ctr}) \quad (20)$$

24 where, in case of $x_{ctr} > 0$, $T_{SMCctr} = s a t (x_{ctr}) = x_{ctr}$;

25 in case of $x_{ctr} \leq 0$, $T_{SMCctr} = s a t (x_{ctr}) = 0$;

$$\begin{aligned}
 &1 \quad x_{ctr} = k_{wctr} \cdot J_w \cdot (d \varepsilon_{ctr} / dt) \\
 &2 \quad + T_{sg} \cdot (s_{ctr} / (|s_{ctr}| + \delta)) \quad (21)
 \end{aligned}$$

$$\begin{aligned}
 &3 \quad \text{where } s_{ctr} = \varepsilon_{ctr} + k_i \cdot \int (\varepsilon_{ctr}) dt \quad (22) \\
 &4 \quad (\text{limits of integral is from 0 to } t);
 \end{aligned}$$

5 k_{wctr} is a differential term gain and in case of $(d \varepsilon_{ctr} / dt) > 0$
 6 is k_{wu} and in case of $(d \varepsilon_{ctr} / dt) \leq 0$, is k_{wd} ; J_w is an inertia
 7 term; T_{sg} is a changing gain; δ is a constant for preventing
 8 chattering; and k_i is an integral term gain.

9 In case where the actual differential rotation speed
 10 $\Delta \omega_{ctr}$ becomes smaller than the control start differential
 11 rotation speed $\Delta \omega_{ctrrs}$, the transfer clutch 21 is engaged in
 12 a static friction condition due to the too small differential
 13 rotation speed and as a result the control of the transfer clutch
 14 21 enters into a slip-lock condition. To prevent this exacerbated
 15 control stability, the clutch torque T_{SMCctr} is established to
 16 0. Further, the integral is reset ($\int (\varepsilon_{ctr}) dt$ is established
 17 to 0). Further, in case where the brake ON signal is inputted,
 18 similarly, the clutch torque T_{SMCctr} is established to 0 in order
 19 to prevent the interference with the braking condition and also
 20 the integral is reset.

21 Then, the establishment of the clutch torque T_{SMCFt}
 22 using the differential rotation speed deviation between front
 23 left and front right wheels ε_{Ft} by the sliding mode will be
 24 described.

$$25 \quad T_{SMCFt} = s a t (x_{Ft}) \quad (23)$$

1 where, in case of $x_{Ft} > 0$, $TSMC_{Ft} = s a t (x_{Ft}) = x_{Ft}$; in case
 2 of $x_{Ft} \leq 0$, $TSMC_{Ft} = s a t (x_{Ft}) = 0$;

$$3 \quad x_{Ft} = k_{wFt} \cdot J_w \cdot (d \varepsilon_{Ft}/dt) \\ 4 \quad + T_{sg} \cdot (s_{Ft}/(|s_{Ft}| + \delta)) \quad (24)$$

$$5 \quad \text{where } s_{Ft} = \varepsilon_{Ft} + k_i \cdot \int (\varepsilon_{Ft}) dt \quad (25)$$

6 (limits of integral is from 0 to t);

7 k_{wFt} is a differential term gain and in case of $(d \varepsilon_{Ft}/dt) > 0$
 8 is k_{wu} and in case of $(d \varepsilon_{Ft}/dt) \leq 0$, is k_{wd} ; J_w is an inertia term;
 9 T_{sg} is a changing gain; δ is a constant for preventing chattering;
 10 and k_i is an integral term gain.

11 In case where the actual differential rotation speed
 12 between front left and front right wheels $\Delta \omega_{Ft}$ becomes smaller
 13 than the control start differential rotation speed $\Delta \omega_{Fts}$, it
 14 is judged that the control of the front differential clutch 27
 15 is effective and the clutch torque $TSMC_{Ft}$ of the transfer clutch
 16 21 is established to 0 to prevent the interference between the
 17 front differential 27 and the transfer clutch 21. Further, the
 18 integral is reset ($\int (\varepsilon_{Ft}) dt$ is established to 0). Further, in
 19 case where the brake ON signal is inputted, similarly, the clutch
 20 torque $TSMC_{Ft}$ is established to 0 in order to prevent the
 21 interference with the braking condition and also the integral
 22 is reset.

23 Then, the establishment of the clutch torque $TSMC_{Rr}$
 24 using the differential rotation speed deviation between rear left
 25 and rear right wheels ε_{Rr} by the sliding mode will be described.

$$1 \quad \text{TSMCRr} = \text{s a t} (x_{Rr}) \quad (26)$$

2 where, in case of $x_{Rr} > 0$, $\text{TSMCRr} = \text{s a t} (x_{Rr}) = x_{Rr}$; in case

3 of $x_{Rr} \leq 0$, $\text{TSMCRr} = \text{s a t} (x_{Rr}) = 0$;

$$4 \quad x_{Rr} = k_{wRr} \cdot J_w \cdot (d \varepsilon_{Rr}/dt) \\ 5 \quad + T_{sg} \cdot (s_{Rr}/(|s_{Rr}| + \delta)) \quad (27)$$

$$6 \quad \text{where } s_{Rr} = \varepsilon_{Rr} + k_i \cdot \int (\varepsilon_{Rr}) dt \quad (28)$$

7 (limits of integral is from 0 to t);

8 k_{wRr} is a differential term gain and, in case of $(d \varepsilon_{Rr}/dt) > 0$,

9 is k_{wu} and, in case of $(d \varepsilon_{Rr}/dt) \leq 0$, is k_{wd} ; J_w is an inertia

10 term; T_{sg} is a changing gain; δ is a constant for preventing

11 chatterings; and k_i is an integral term gain.

12 In case where the actual differential rotation speed
13 between rear left and rear right wheels $\Delta \omega_{Rr}$ becomes smaller
14 than the control start differential rotation speed $\Delta \omega_{Rrs}$, it
15 is judged that the control of the front differential clutch 27
16 is effective and the clutch torque TSMCRr of the transfer clutch
17 21 is established to 0 to prevent the interference between the
18 front differential 27 and the transfer clutch 21. Further, the
19 integral is reset ($\int (\varepsilon_{Rr}) dt$ is established to 0). Further, in
20 case where the brake ON signal is inputted, similarly, the clutch
21 torque TSMCRr is established to 0 in order to prevent the
22 interference with the braking condition and also the integral
23 is reset.

24 Thus, according to the sliding mode control in the
25 embodiment, the switching function is formed using the polarity

of the integral term of deviation. That is, in the switching function (18), the integral term of deviation $sctrf$ is divided by $(|sctrf| + \delta)$ to obtain the polarity of the integral term and in the changing function (21), the integral term of deviation $sctrr$ is divided by $(|sctrr| + \delta)$ to obtain the polarity of the integral term, in the changing function (24), the integral term of deviation sFt is divided by $(|sFt| + \delta)$ to obtain the polarity of the integral term, and in the changing function (27), the integral term of deviation sRr is divided by $(|sRr| + \delta)$ to obtain the polarity of the integral term. In these cases, δ is a value for preventing the division by 0. Hence, even in case where the respective integral terms are small, since the clutch torque is established by applying the integral terms to the sliding mode control, the control according to the present invention provides a traction performance with accurate and quick responsibility.

The respective clutch torques $TSMCctrf$, $TSMCctrr$, $TSMCFt$, $TSMCRr$ thus calculated in the sliding mode control clutch torque calculating section 52g are outputted to the feedback control clutch torque calculating section 52i.

The deviation proportional control clutch torque calculating section 52h inputs the brake signal processed in the brake switch delaying section 52a, the actual differential rotation speed between front and rear drive shafts $\Delta \omega ctrf$, $\Delta \omega ctrr$ from the actual front-rear rotation speed calculating section 52b, the actual differential rotation speed between the

1 front left wheel 14FL and the front right wheel 14FR differential
 2 rotation speed $\Delta \omega_{Ft}$ from the actual front left-right differential
 3 rotation speed calculating section 52c, the actual differential
 4 rotation speed between the rear left wheel 14RL and the rear right
 5 wheel 14RR differential rotation speed $\Delta \omega_{Rr}$ from the actual rear
 6 left-right differential rotation speed calculating section 52d,
 7 the respective control start differential rotation speeds $\Delta \omega_{ctrfs}$,
 8 $\Delta \omega_{ctr rs}$, $\Delta \omega_{Fts}$, $\Delta \omega_{Rrs}$ from the control start
 9 differential rotation speed establishing section 52e, and the
 10 respective target differential rotation speeds $\Delta \omega_{ctrft}$, $\Delta \omega_{ctr rt}$,
 11 $\Delta \omega_{Ftt}$, $\Delta \omega_{Rrt}$ from the target differential rotation speed
 12 establishing section 52f. Further, this deviation proportional
 13 control clutch torque calculating section 52h calculates the
 14 deviation between the target differential rotation speed and the
 15 actual differential rotation speed for respective rotation speeds
 16 as will be described hereinafter and obtains proportional
 17 components of the clutch torques for converging the actual
 18 differential rotation speed upon the target differential rotation
 19 speed as follows (clutch torques T_{pcctrf} , $T_{pcctr r}$, T_{pcFt} , T_{pcRr}).

20 That is, the deviation between the target differential
 21 rotation speed and the actual differential rotation speed for
 22 the respective rotation speeds is calculated as follows:

$$\begin{aligned}
 \varepsilon_{pctrf} &= \Delta \omega_{ctrf} - \Delta \omega_{ctrft} \\
 &\quad - (\Delta \omega_{ctrft} - \Delta \omega_{ctrfs}) \quad (29)
 \end{aligned}$$

$$\varepsilon_{pctr r} = \Delta \omega_{ctr r} - \Delta \omega_{ctr rt}$$

$$1 \quad \quad \quad - (\Delta \omega_{ctr rt} - \Delta \omega_{ctr rs} \quad (30))$$

$$2 \quad \varepsilon_{pFt} = \Delta \omega_{Ft} - \Delta \omega_{Ftt}$$

$$3 \quad \quad \quad - (\Delta \omega_{Ftt} - \Delta \omega_{Fts}) \quad (31)$$

$$4 \quad \varepsilon_{pRr} = \Delta \omega_{Rr} - \Delta \omega_{Rrt}$$

$$5 \quad \quad \quad - (\Delta \omega_{Rrt} - \Delta \omega_{Rrs}) \quad (32)$$

6 where ε_{pctrf} is a differential rotation speed deviation between
7 front and rear drive shafts; $\varepsilon_{pctr r}$ is a differential rotation
8 speed deviation between front and rear drive shafts; ε_{pFt} is a
9 differential rotation speed deviation between front left and front
10 right wheels; and ε_{pRr} is a differential rotation speed deviation
11 between rear left and rear right wheels.

12 The clutch torques T_{pcctrf} , $T_{pcctr r}$, T_{pcFt} , T_{pcRr} based
13 on the deviation proportional control are calculated respectively
14 as follows:

15 First, the clutch torque T_{pcctrf} based on the deviation
16 proportional control using the differential rotation speed
17 deviation between front and rear shafts ε_{pctrf} is,

18 in case of $\varepsilon_{pctrf} > 0$, $T_{pcctrf} = k_{p1} \cdot \varepsilon_{pctrf} + k_{p2} \cdot \Delta \omega_{ctrf}$

19 in case of $\varepsilon_{pctrf} \leq 0$, $T_{pcctrf} = k_{p2} \cdot \Delta \omega_{ctrf}$.

20 Next, the clutch torque $T_{pcctr r}$ based on the deviation
21 proportional control using the differential rotation speed
22 deviation between front and rear shafts $\varepsilon_{pctr r}$ is,

23 in case of $\varepsilon_{pctr r} > 0$, $T_{pcctr r} = k_{p1} \cdot \varepsilon_{pctr r} + k_{p2} \cdot \Delta \omega_{ctr r}$

24 in case of $\varepsilon_{pctr r} \leq 0$, $T_{pcctr r} = k_{p2} \cdot \Delta \omega_{ctr r}$.

25 Next, the clutch torque T_{pcFt} based on the deviation

1 proportional control using the differential rotation speed
 2 deviation between front left and front right ε_{pFt} is,
 3 in case of $\varepsilon_{pFt} > 0$, $T_{pcFt} = k_{p1} \cdot \varepsilon_{pcFt} + \Delta \omega_{cFt}$
 4 in case of $\varepsilon_{pFt} \leq 0$, $T_{pcFt} = \Delta \omega_{Ft}$.

5 Next, the clutch torque T_{pcRr} based on the deviation
 6 proportional control using the differential rotation speed
 7 deviation between rear left and rear right ε_{pRr} is,
 8 in case of $\varepsilon_{pRr} > 0$, $T_{pcRr} = k_{p1} \cdot \varepsilon_{pcRr} + \Delta \omega_{cRr}$
 9 in case of $\varepsilon_{pRr} \leq 0$, $T_{pcRr} = \Delta \omega_{Rr}$.

10 where k_{p1} is a first proportional term gain; k_{p2} is a second
 11 proportional term gain; ε_{ctrf} is a differential rotation speed
 12 deviation between front and rear drive shafts; ε_{ctrr} is a
 13 differential rotation speed deviation between front and rear drive
 14 shafts; ε_{Ft} is a differential rotation speed deviation between
 15 front left and front right wheels; and ε_{Rr} is a differential
 16 rotation speed deviation between rear left and rear right wheels.

17 Further, when the ON signal of the brake switch is
 18 inputted, the aforesaid clutch torques T_{pcctrf} , T_{pcctrr} , T_{pcFt} ,
 19 T_{pcRr} based on the deviation proportional control are established
 20 to 0 to avoid the interference with the braking condition,
 21 respectively.

22 The clutch torques T_{pcctrf} , T_{pcctrr} , T_{pcFt} , T_{pcRr}
 23 calculated in the deviation proportional control clutch torque
 24 calculating section 50j are outputted to the feedback control
 25 clutch torque calculating and outputting section 52i,

1 respectively.

2 The feedback control clutch torque calculating and
3 outputting section 52i inputs the respective clutch torques
4 TSMCctrf, TSMCctrr, TSMCFt, TSMCRr from the sliding mode control
5 clutch torque calculating section 52g and the respective clutch
6 torques Tpcctrf, Tpcctrr, TpcFt, TpcRr from the deviation
7 proportional control clutch torque calculating section 52h.

8 Then, four clutch torques Tctrf, Tctrr, TFt, TRr are
9 obtained by the summation respectively and a largest one of the
10 clutch torques is established to be a final clutch torque Tlsdfb
11 to be applied to the transfer clutch 21. When an execution command
12 is issued from the mode establishing section 54, this clutch torque
13 Tlsdfb is outputted to the clutch torque calculating section 55.

14 That is, $Tctrf = TSMCctrf + Tpcctrf$

15 $Tctrr = TSMCctrr + Tpcctrr$

16 $TFt = TSMCFt + TpcFt$

17 $TRr = TSMCRr + TpcRr$

18 $TLsdfb = \text{MAX} (Tctrf, Tctrr, TFt, TRr)$ (33)

19 Thus, the sliding mode control clutch torque calculating section
20 52g, the deviation proportional control clutch torque calculating
21 section 52b and the feedback control clutch torque calculating
22 and outputting section 52i, serve as clutch torque calculating
23 and outputting means in the automatic clutch control means.

24 On the other hand, as shown in Fig. 2, the manual mode
25 control section 53 inputs a signal indicative of the dial position

1 from the characteristic changing dial 36 and a signal indicative
2 of an output execution command from the mode establishing section
3 54.

4 The manual mode control section 53 acts as manual clutch
5 control means in which, when the output execution command of the
6 calculated value is inputted from the mode establishing section
7 54, a clutch torque T_{lsdh} corresponding to the dial position
8 selected by the driver of the characteristic changing dial 36
9 is outputted to the clutch torque calculating section 55.

10 The mode establishing section 54 serving as control
11 selecting means inputs an ON-OFF signal of the ignition switch
12 34, a signal indicative of the selection of the power distribution
13 control mode (automatic mode or manual mode) from the mode switch
14 35 and the vehicle speed V from the vehicle speed calculating
15 section 51.

16 Further, the mode establishing section 54 outputs an
17 output execution command of calculated values to the automatic
18 mode calculating section 52 or the manual mode control section
19 53 according to the signal from the mode switch 35. In the following
20 two cases, the output execution command of the calculated values
21 is issued to the automatic mode control section 52. The mode
22 established in the mode establishing section 54 is indicated by
23 the mode indicator lamp 37.

24 [Case 1] In an initial condition of the ignition switch 34 turned
25 on, even in case where the manual mode is selected by the mode

1 switch 35, the output execution command of the calculated values
2 is outputted to the automatic mode control section 52 until the
3 manual mode is newly selected by the mode switch 35. Accordingly,
4 in case where a driver turns the ignition switch 35 off and leaves
5 the car while the manual mode is selected, when the car starts
6 next, the automatic mode is designed to be securely selected so
7 as to obtain an optimum clutch torque.

8 [Case II] In case where the vehicle travels at a speed exceeding
9 a threshold value V_H (for example, 50 km/hour), even if the manual
10 mode is selected by the mode switch 35, the output execution command
11 of the calculated values is outputted to the automatic mode control
12 section 52 and the automatic mode is selected. This is because
13 it is judged that the optimum clutch engagement force is difficult
14 to be obtained in the manual mode condition under such a high
15 speed condition.

16 The clutch torque calculating section 55 inputs the
17 ON-OFF signal from the brake switch 33, the ON-OFF signal from
18 the ABS control apparatus 38, the clutch torque T_{lsdfb} from the
19 automatic mode control section 52 (when the output execution
20 command is issued from the mode establishing section 54), and
21 the clutch torque T_{lsdh} from the manual mode control section 53
22 (when the output execution command is issued from the mode
23 establishing section 54), respectively.

24 These inputted clutch torques T_{lsdfb} or T_{lsdh} are
25 converted into a signal of a clutch torque T_{lsd} and outputted

1 to the center differential drive section 60.

2 When a turned-on signal is inputted from the ABS control
3 apparatus 38, that is, when there is a signal indicating that
4 the ABS is operative, the clutch torque T_{lsd} is established to
5 a predetermined constant value C_{ABS} in order to prevent the
6 interference with the ABS control. Similarly, when a turned-on
7 signal is inputted from the brake switch 33, the clutch torque
8 T_{lsd} is established to a predetermined constant value C_{brk} in
9 order to prevent the interference with the braking condition.

10 Next, the front-rear power distribution control of the
11 differential limiting section 50 will be described by referring
12 to a flowchart of Fig. 6. This flowchart is a flowchart which
13 will be executed at a specified time interval when the ignition
14 switch 34 is turned on. First, at S101, parameters are read and
15 then at S102 it is judged whether or not the ignition switch 34
16 is still in a turned-on condition.

17 In case where it is judged at S102 that the ignition
18 switch 34 is turned on, the program again returns to S101, and
19 in case where it is judged that the ignition switch 34 is turned
20 off, the program goes to S103.

21 At S103, it is judged whether or not a signal is inputted
22 from the mode switch 35. In case where no signal is inputted,
23 the program goes to S104 where the front-rear power distribution
24 control is established to the automatic mode and the mode indicator
25 lamp 37 indicates the automatic mode.

1 Then, the program goes to S105 where the mode
2 establishing section 54 outputs the output execution command of
3 the calculated values to the automatic mode control section 52
4 and the clutch torque T_{lsdfb} calculated in the automatic mode
5 control section 52 is outputted to the clutch torque calculating
6 section 55. The clutch torque calculating section 55 outputs the
7 inputted clutch torque T_{lsdfb} according to the ON-OFF conditions
8 of the brake switch 33 or the ABS control apparatus 38.

9 On the other hand, at S103, in case where the signal
10 is inputted from the mode switch 35, the program goes to S106
11 where it is judged whether or not the front-rear power distribution
12 control selected by the mode switch 35 is the manual mode. As
13 a result of the judgment, if the power distribution control selected
14 is not the manual mode, it must be the automatic mode and the
15 program goes to S104. If it is the manual mode, the program goes
16 to S107.

17 At S107, it is judged whether or not the present vehicle
18 speed V exceeds the preestablished threshold value V_H , for example
19 50 km/hour, ($V > V_H$). In case of $V > V_H$, it is judged that an optimum
20 clutch torque is difficult to be obtained with the clutch torque
21 established by the driver and the program goes to S104 where the
22 front-rear power distribution control is established to the
23 automatic mode.

24 In case where the present vehicle speed is lower than
25 the threshold value V_H ($V \leq V_H$), the program goes to S108 where

1 the front-rear power distribution control is established to the
2 manual mode and the mode indicator lamp 37 indicates as such.

3 Then, the program goes to S109 where the mode
4 establishing section 54 outputs the output execution command of
5 the clutch torque established by the driver to the manual mode
6 control section 53. The manual mode control section 53 outputs
7 the clutch torque Tlsdh to the clutch torque calculating section
8 55, from which the inputted clutch torque Tlsdh is outputted
9 according to the ON-OFF conditions of the ABS control apparatus
10 38 or the brake switch 33.

11 According to the embodiment of the present invention,
12 when the ignition switch 34 is at an initial stage of a turned-on
13 condition, the front-rear power distribution control is designed
14 to be established to the automatic mode until the manual mode
15 is newly selected by the mode switch 35. Hence, in case where
16 the driver turns the ignition switch 35 off with the manual mode
17 selected and leaves the vehicle, when the driver starts the vehicle
18 next, the automatic mode is securely selected. As a result, even
19 in case where the driver starts the vehicle inadvertently with
20 the manual mode, the automatic mode is automatically selected
21 and as a result such an unexpected vehicle behavior as the vehicle
22 encounters spin conditions on a road surface with low friction
23 coefficient, can be prevented. Further, according to the
24 embodiment of the present invention, since the power distribution
25 control is forcedly established to the automatic mode when the

1 vehicle speed V exceeds the threshold value V_H , always the optimum
2 clutch torque can be obtained. This prevents the vehicle behavior
3 from becoming unstable due to unsuitable clutch torques selected
4 by the manual mode. Further, since the power distribution control
5 enters into the automatic mode automatically when the vehicle
6 speed exceeds the threshold value, for example 50 km/hour, the
7 fuel economy is prevented from being exacerbated by a long time
8 engagement of the transfer clutch 21.

9 The entire contents of Japanese Patent Application No.
10 Tokugan 2002-311568 filed October 25, 2002, is incorporated
11 herein by reference.

12 While the present invention has been disclosed in terms
13 of the preferred embodiment in order to facilitate better
14 understanding of the invention, it should be appreciated that
15 the invention can be embodied in various ways without departing
16 from the principle of the invention. Therefore, the invention
17 should be understood to include all possible embodiments which
18 can be embodied without departing from the principle of the
19 invention set out in the appended claims.

20